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Influence of Thermal Boundary Conditions on Heat Transfer From a Cylinder in Crossflow

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Summary

Local heat transfer data over the leading surface of a cylinder in crossflow were obtained for a Reynolds number range of 50 000 to 215 000. The cylinder was operated at both uniform-wall-temperature and uniform-heat-flux thermal boundary conditions. At an angular distance of 80° from the front stagnation point the uniform-wall-temperature heat transfer coefficients were as much as 66 percent lower than the uniform-heat-flux data.

Between the stagnation point and 60° around the cylinder there were no significant differences in the data. This region of the cylinder is well within the cylindrical curvature region of the front end of a real turbine blade. It was therefore concluded that either thermal boundary condition could be used to model turbine flow over that region of the blade.

Also presented herein are the results of evaluating the exponent x in the fundamental relationship $\mathrm{Nu}=f(\mathrm{Re})^x$, which is used in data correlation. The exponent varied as a function of local position on the cylinder even in the laminar flow region. The value of x increased linearly from 0.50 at the stagnation point to 0.59 at 60° around the cylinder. This linear trend continued into the separation region at 80° for the uniform-wall-temperature data, but x increased markedly in the separation region for the uniform-heat-flux data.

Introduction

In modeling flow through a turbine a cylinder in crossflow is usually used to represent flow over the front edge of the turbine blade. By generating a heat flux in the walls of the cylinder and properly modeling the flow and geometry parameters, it is possible to obtain heat transfer coefficients that have design application. The heat transfer data accumulated in this manner have been obtained under different thermal boundary conditions, and in recent years there have been divergent opinions expressed in the literature on the extent of this boundary condition influence. The opinions ranged from insignificant thermal boundary condition effects (ref. 1) to large effects (ref. 2) of the order of 30 percent. Theoretical analyses for the laminar flow region (refs. 2 and 3) predict significant differences in the data because of these boundary conditions.

The thermal boundary conditions used for calculating heat transfer coefficients have been more often than not just a design consideration for a particular study. The basic equation used is

$$q=h(T_w-T_\infty) \tag{1}$$

where q is heat flux, h is heat transfer coefficient, T_w is cylinder wall temperature, and T_∞ is free-stream temperature. With a known free-stream temperature the heat transfer coefficient around the cylinder varies with the wall temperature if the heat flux is held constant and varies with the heat flux if the wall temperature is held constant. Consequently the heat transfer coefficient can be calculated with either one of these thermal boundary conditions.

In the past, test cylinders were uniquely designed with control of the power input to produce either a uniform heat flux or a uniform wall temperature. In some cases the heat flux was not that well defined. In either case data comparisons under different thermal conditions could only be made with data from different test facilities. Consequently the influence of thermal boundary conditions alone could not be isolated because system heat losses, flow conditions, tunnel characteristics, etc., were unique for each test facility. In attempting to make data comparisons some investigators have attributed differences in the data exclusively to the thermal boundary conditions, but for example they did not—or could not—include the influence of upstream turbulence conditions (refs. 2, 4, and 5), which can be quite significant.

The present study circumvents these problems of data comparison by using a single test facility and a cylinder designed with control of the power input to generate heat fluxes with either uniform-heat-flux or uniform-wall-temperature boundary conditions. The heat transfer data presented herein were obtained from the front stagnation point to an angular distance of 100° around the circumference of the cylinder. The cylindrical portion of the front end of a real turbine blade is well within this region on the test cylinder, which is therefore of interest for turbine blade modeling.

Apparatus

A schematic drawing of the test cylinder (made partly out of a hard, copper, thick-wall tube with an outside diameter of 4.82 cm, a wall thickness of 0.38 cm, and a length of 15.24 cm) is shown in figure 1. The heat transfer surface was made of half the copper tube cut longitudinally and backed with a

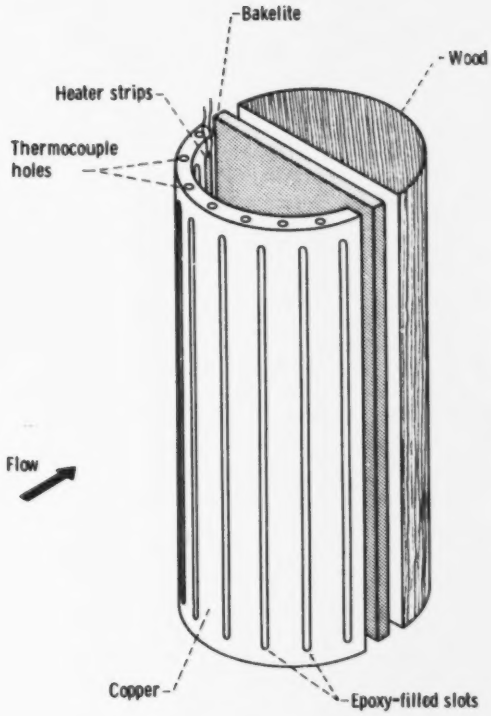


Figure 1. - Cylinder components.

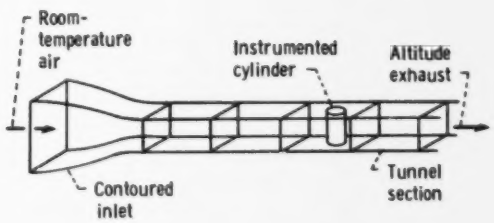


Figure 2. - Schematic of tunnel.

phenolic plastic plate. The rear half of the cylinder was made of wood.

The outer surface of the copper half-cylinder was divided into nine equal-area sections by cutting longitudinal slots 0.20 centimeter wide through the tube at 20° intervals on the circumference. The slots were then filled with a low-thermal-conductivity epoxy to minimize circumferential heat transfer.

The divided copper strips were instrumented with Chromel-Alumel thermocouples pressure mounted with springs in holes drilled through the end wall to the center of the cylinder. Thermofoil heaters were installed behind each copper strip with pressure-sensitive adhesive. The void volume in the half-cylinder was filled with fiberglass insulation, and the ends were capped with phenolic plastic plates.

The test cylinder was mounted in a rectangular tunnel (fig. 2) having a 15.24- by 38.10-centimeter flow-area cross section, a contoured inlet, and a transition piece that connects to the laboratory altitude exhaust system.

Experimental Procedure

The cylinder was mounted in the tunnel with the stagnation point at the second copper strip from the end of the instrumented half-cylinder (fig. 1). Heater strips on both ends of the half-cylinder were then used as guard heaters to minimize heat losses. The controlled parameters for these tests were tunnel air velocity and power to the individual heater strips. Two sets of data were obtained at each tunnel velocity. By proper control of each heater strip comparative sets of data were obtained at uniform wall temperature and at uniform heat flux. In both cases sufficient running time was allowed for equilibrium conditions to be established.

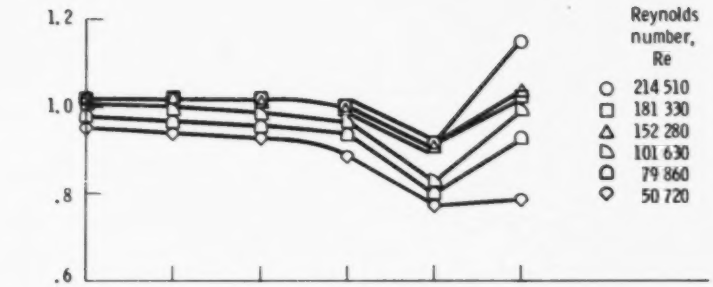
No attempt was made to account for other heat losses, such as losses through thermocouple wires and end walls and internal losses to the cylinder, since it was assumed that they were approximately the same under both thermal boundary conditions. The absolute values of the data were not of primary interest but rather the accuracy of the relative differences between the uniform-heat-flux and uniform-wall-temperature data.

Data and Results

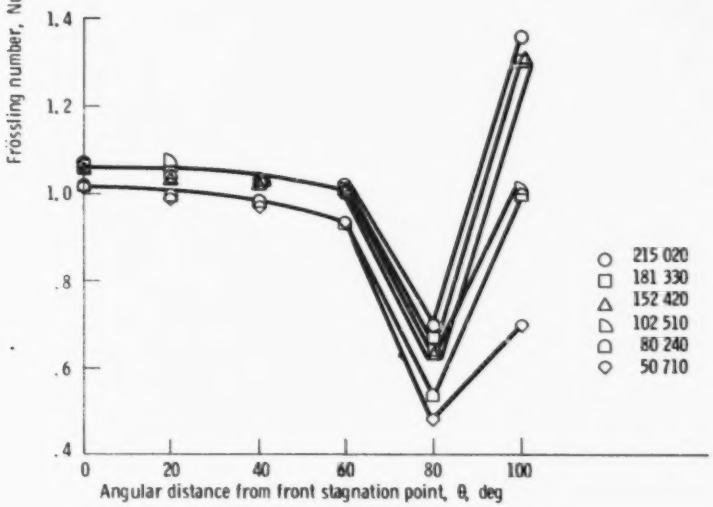
Six sets of data were obtained over a range of cylinder Reynolds number Re (based on diameter) from 50 000 to 215 000. The temperature difference between the cylinder surface and the tunnel air ranged from 23 to 44 degrees celsius, with an average heat flux of about 5000 watts per square meter.

Local heat transfer data in terms of Frössling number $Nu/\sqrt{Re}$ are plotted in figure 3 from the front stagnation point to 100° around the circumference of the cylinder. Figure 3(a) presents the uniform-heat-flux data, and figure 3(b) presents the uniform-wall-temperature data.

In both cases the heat transfer maximized at the front stagnation point and decreased to a minimum at an angular distance around the cylinder of about 80°, which is in the laminar separation region. Between 80° and 100° on the circumference of the



(a) At uniform-heat-flux conditions.



(b) At uniform-wall-temperature conditions.

Figure 3. - Local heat transfer.

cylinder the flow separates, and the separation was followed by a rapid increase in heat transfer. This report compares heat transfer data from the front stagnation point to an angular distance of 80° around the cylinder.

An example of the data comparison at a Reynolds number of about 80 000 is presented in figure 4. Significant differences lie within the separation region, where the uniform-heat-flux data were 50 percent greater than the uniform-wall-temperature data. This data trend persists over the entire range of Reynolds numbers covered in this study.

The present uniform-heat-flux data are compared with some representative data from the literature (refs. 6 to 8) in figure 5. The four sets of data on this plot lie within a Reynolds number range of about 100 000 to 126 000. At the stagnation point all five curves plot within a narrow range of scatter, but significant differences occur in the separation region. At 80° around the cylinder the present data curve

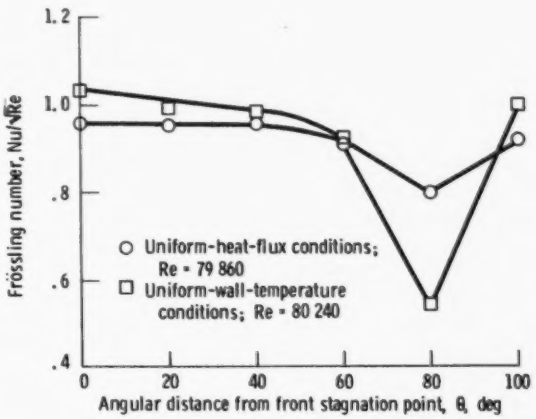


Figure 4. - Comparison of local heat transfer data at uniform-heat-flux and uniform-wall-temperature conditions.

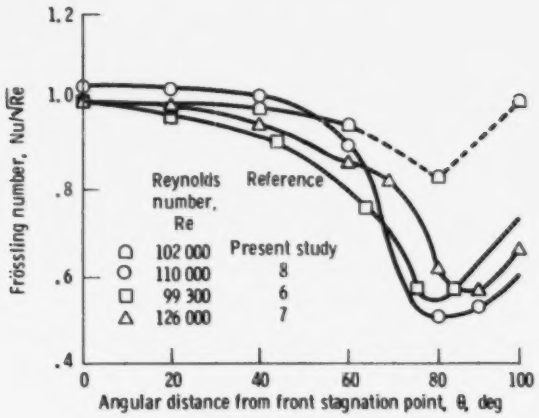


Figure 5. - Comparison of Frössling number profile with reference data for uniform-heat-flux thermal boundary condition data.

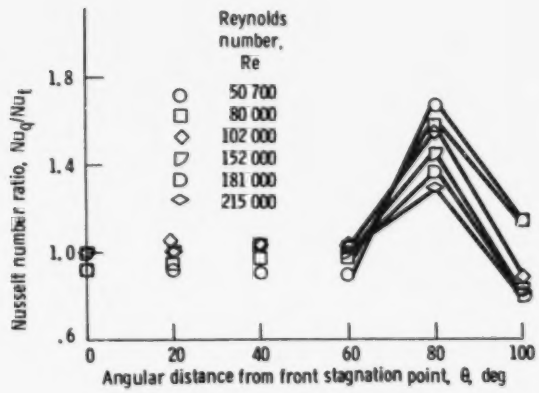


Figure 6. - Local Nusselt number ratio based on uniform-heat-flux and uniform-wall-temperature conditions as a function of angular distance.

does not fall off as far as the data from the references. It is suggested that the relatively high Nusselt number at the separation point for the present data is caused by the cylinder design. The epoxy-filled slots (fig. 1) left unheated gaps on the surface of the cylinder and delayed the growth of the thermal layer.

The ratio of Nusselt numbers based on uniform heat flux and uniform wall temperature Nu_q/Nu_t as a function of angular distance from the front stagnation point θ is presented in figure 6. Over the Reynolds number range covered (50 700 to 215 000) significant differences in the data only occur at angular distances larger that 60° from the front stagnation point. At $\theta = 80°$, within the separation region, these differences range from 28 percent at the higher Reynolds numbers to 66 percent at the lower Reynolds numbers. For $\theta \leq 60°$ average differences of about 5 percent are noted in the data only at low Reynolds numbers (50 700 and 80 000).

The Nusselt number-Reynolds number relationships for both sets of data were then determined from the slopes of the lines drawn through the data points on log-log plots (fig. 7). The data were averaged at the low Reynolds numbers. Each figure presents data for a particular position on the cylinder from 0° to 80°, at 20° intervals. At the front stagnation point (fig. 7(a)) the exponent on the Reynolds number is 0.50, which agrees with values reported in the literature. On the succeeding figures ($\theta = 20°$ (fig. 7(b)) to $\theta = 60°$ (fig. 7(d)) which is still in the laminar flow region the exponent increases to

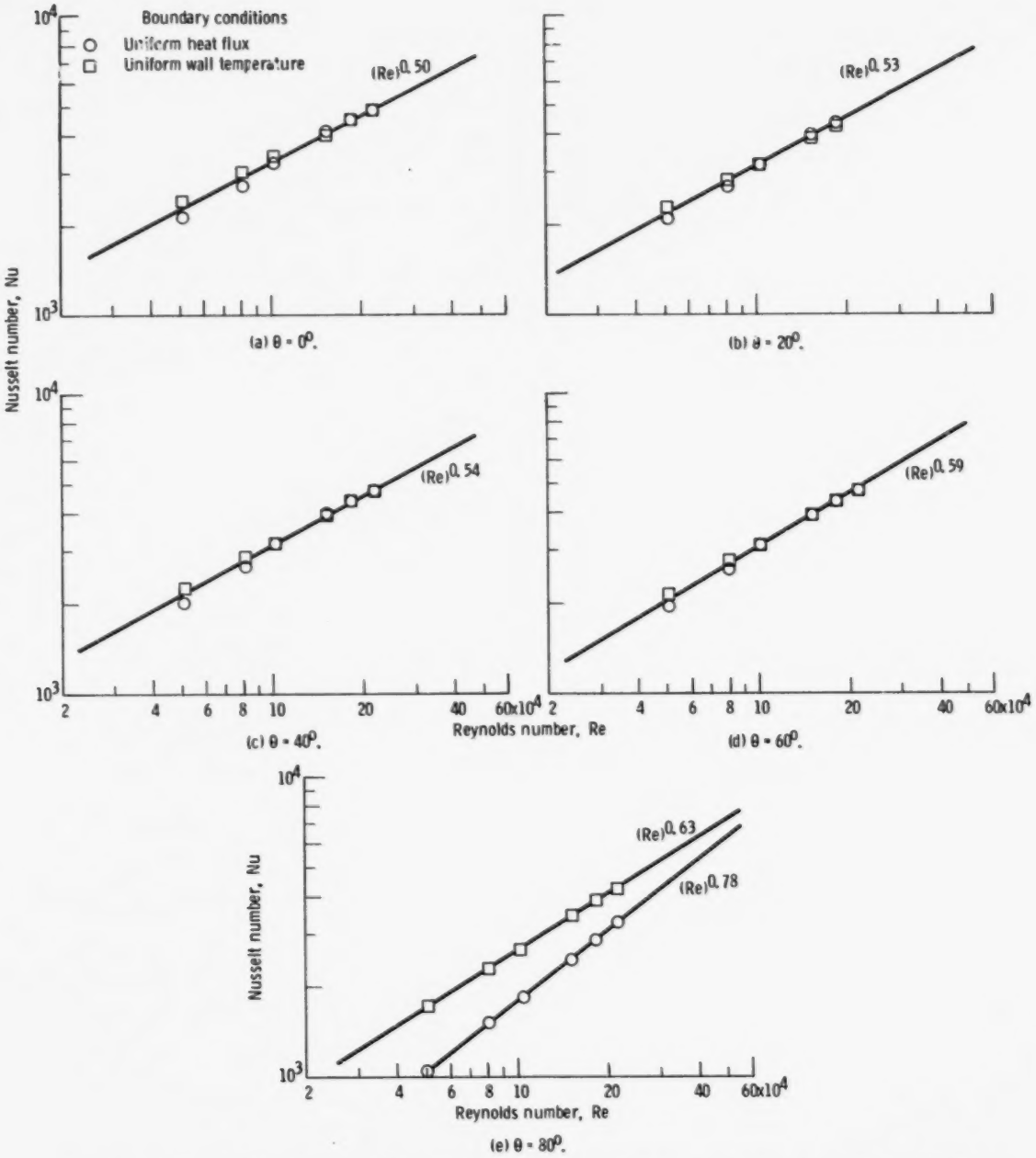


Figure 7. - Relationship between Nusselt and Reynolds numbers based on uniform-heat-flux and uniform-wall-temperature thermal boundary conditions at various angular distances from front stagnation point θ.

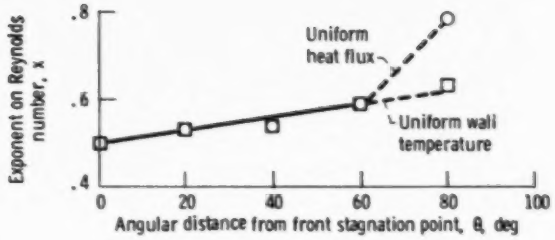


Figure 8. - Variation of exponent x in relationship $Nu = f(Re)^x$ as a function of position on cylinder for both uniform-heat-flux and uniform-wall-temperature thermal boundary conditions.

0.59. In the flow separation region ($\theta = 80°$, fig. 7(e)) the data separate into two distinct curves with the exponent for the uniform-heat-flux case greater than that for the uniform-wall-temperature case.

These results are summarized in figure 8, which shows the exponent x as a function of θ for the data obtained under the two different thermal boundary conditions. The exponent varies from 0.50 at the stagnation point to 0.59 at $\theta = 60°$. This change of exponent for laminar boundary layer flow over a cylinder has not been reported in the literature. For example, design equations for the leading-edge region of the turbine blade, such as those reported in reference 9, use the stagnation-point value of $x = 0.50$ over the entire cylindrical portion of the blade.

In the separation region ($\theta = 80°$) figure 8 shows that the exponents for the uniform-temperature case still follow the linear increase as observed in the laminar region but that the exponent for the uniform-heat-flux case markedly increases in value, departing from linearity.

Summary of Results

The influence of thermal boundary conditions on heat transfer from a cylinder in crossflow has been discussed in the literature, but experimental verification has been confined to comparison of data from different test facilities. Under these conditions some doubt remains as to the validity of these comparisons because of heat losses, upstream turbulence flow conditions, and tunnel geometries peculiar to each test facility.

The present study minimizes this doubt by using a single test facility and a cylinder designed with the capability of generating heat fluxes under both uniform-heat-flux and uniform-wall-temperature thermal boundary conditions. Data differences could therefore be attributed solely to the influence of the thermal boundary conditions. Two sets of data were obtained under each thermal boundary condition over a Reynolds number range of 50 000 to 215 000.

In the flow separation region at angular distances from the front stagnation point greater than 60°, the uniform-wall-temperature heat transfer coefficients were as much as 66 percent lower than the uniform-heat-flux coefficients.

In the laminar flow region between the stagnation point and up to 60° around the cylinder, there were no significant differences in the heat transfer data. Consequently, since cylindrical curvature on the leading edge of a real turbine blade is less than 60°, either thermal boundary condition could be assumed in predicting local heat transfer rates when using a cylinder to model turbine flow.

For the fundamental relationship $Nu = f(Re)^x$, widely used in data correlation, the data reported herein show the exponent x to be a function of position on the cylinder. In the laminar flow region, for both sets of data, the value of x increased linearly from 0.50 at the stagnation point to 0.59 at 60° around the cylinder. This trend continued into the separation region at 80° for the uniform-wall-temperature data, but x increased markedly in this region for the uniform-heat-flux data.

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio, December 23, 1980

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15. Supplementary Notes

16. Abstract

Local heat transfer data over the leading surface of a cylinder in crossflow were obtained for a Reynolds number range of 50 000 to 215 000. The cylinder was operated at both uniform-wall-temperature and uniform-heat-flux thermal boundary conditions. At an angular distance of 80^{o} from the front stagnation point the uniform-wall-temperature heat transfer coefficients were as much as 66 percent lower than the uniform-heat-flux data. Between the stagnation point and 60^{o} around the cylinder there were no significant differences in the data. This region of the cylinder is well within the cylindrical curvature region of the front end of a real turbine blade. It was therefore concluded that either thermal boundary condition could be used to model turbine flow over that region of the blade. Also presented herein are the results of evaluating the exponent x in the fundamental relationship $Nu=f(Re)^{x}$, which is used in data correlation. The exponent varied as a function of local position on the cylinder even in the laminar flow region. The value of x increased linearly from 0.50 at the stagnation point to 0.59 at 60^{o} around the cylinder. This linear trend continued into the separation region at 80^{o} for the uniform-wall-temperature data, but x increased markedly in the separation region for the uniform-heat-flux data.

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